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VARIABLE CONDUCTANCE HEAT PIPE

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VARIABLE CONDUCTANCE HEAT PIPE

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ABSTRACT

A passive feedback system, which stabilizes the heat-source temperature (T_s) of a gas-loaded heat pipe, was designed and tested. The control of T_s is accomplished by an auxiliary liquid that senses the heat source and actuates a metal bellows system due to the liquid's thermal expansion. The movement of the bellows varies the gas reservoir volume and leads to a corresponding change of the condensation area of the heat pipe. With methanol as the heat-pipe working fluid and perfluoro-n-pentane as the auxiliary liquid, the control capability was found to be $T_s = 31.5 \pm 1.5^\circ\text{C}$ in a power range from 3 to 30 W, compared to $T_s = 33 \pm 3^\circ\text{C}$ with methanol as auxiliary liquid. The change in T_s was $35 \pm 5.5^\circ\text{C}$ with the bellows held in the closed position.

INTRODUCTION

The conventional gas-controlled variable conductance heat pipe is able to maintain its own temperature nearly constant while heat input or environmental conditions are changing. In many applications, however, the heat-source temperature rather than the heat-pipe temperature has to be stabilized. To a certain extent this can be accomplished by a conventional gas-loaded heat pipe, but only in cases with low thermal resistance between heat source and heat pipe. If the thermal impedance is not negligible, the variations in source temperature can be reduced significantly by the use of a passive or an active feedback system.

The passive system uses a variable volume reservoir. An increase in heat-source temperature causes an auxiliary bellows to expand, which in turn expands the reservoir, and a consequent movement of the vapor/gas front takes place.

In the active system, the heat source is monitored electrically and the signals drive a small heater on the fixed gas reservoir which, in turn, varies the partial pressure of the working fluid contained therein. The obtained gas displacement from or into the reservoir then causes a shift of the vapor/gas front.

A recent feasibility study¹ concludes that both active and passive feedback-controlled, variable conductance heat pipes are feasible. However, the requirement of an external electronic equipment and power supply, and the

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need to cool the reservoir significantly below the heat pipe's temperature at maximum power condition, may limit the general application of the active system. It is therefore desirable to learn more about the performance characteristics of the passive feedback system.

A first attempt to build and test a passive feedback-controlled heat pipe was made under NASA contract by the University of Washington.² Although the feasibility of passive feedback control was demonstrated, the improvement in control sensitivity compared with that obtained with a fixed volume reservoir of the same test configuration was not impressive. Apparently, a major problem was the susceptibility to failure of the bellows system by overpressurization and fatigue.

PRINCIPLE OF OPERATION

Figure 1 is a schematic sketch of a passive feedback-controlled heat pipe. The control system consists of at least two bellows and a sensing bulb located near the heat source. The inner bellows contains an auxiliary incompressible liquid and is connected to the sensing bulb by a capillary tube. The outer bellows is used as a variable storage volume for the noncondensable gas. Variation in the source temperature will cause a change in the pressure of the auxiliary liquid, which results in a displacement of the inner bellows and therefore of the gas storage bellows and of the vapor-gas interface in the heat pipe.

A desired performance of a passive controlled heat pipe is shown in Fig. 2. In this example, the increase of the source temperature T_s is only 2°C over an input power range from 5 to 30 W, while the heat-pipe temperature T_v decreases from 24.5° to 14°C at constant sink condition. The thermal resistance between heat source and heat pipe is considered to be $r = 0.5^\circ\text{C/W}$.

The decrease of the heat-pipe temperature T_v during increase of the power input has two major negative effects on the control sensitivity, which do not appear in conventional gas-loaded heat pipes:

- The temperature difference and the specific heat flow between condenser and heat sink decrease with increasing heat load, which has to be compensated by additional condenser area, i.e., movement of the bellows system.
- The heat-pipe pressure decreases and therefore the gas volume expands, which again has to be compensated by a corresponding movement of the bellows system.

To obtain close control of the source temperature, a large variation of the gas storage volume is therefore required.

DESIGN CONSIDERATIONS

Due to the difficulty in covering the convolutions of the bellows with a wick, passive feedback-controlled heat pipes are considered to be hot (or dry) reservoir heat pipes, with the option to couple the reservoir thermally to the evaporator. This option leads to the basic concept of the presented design, in which the heat source, auxiliary liquid reservoir, heat pipe

evaporator, and bellows system form an integral part (Fig. 3). The gas reservoir is, in this case, exposed to the known variations of the evaporator temperature rather than to the sink condition, and the fact that the evaporator temperature decreases, while the heat load increases, improves the control mechanism since the cooler gas occupies a smaller volume when a large condenser area is needed.

Because of the required large effective area of the gas storage bellows, a small pressure difference between inside and outside the bellows may cause high axial forces that have to be compensated by an adequate spring rate. However, the stiffness of this high spring rate bellows is undesirable because the auxiliary liquid bellows works against it, causing an undue pressure within the incompressible liquid. In the presented design, the gas storage bellows is therefore surrounded with a container that is filled with a small amount of heat-pipe working liquid. The container will then act as a heat pipe and, by providing a good thermal contact between container and evaporator, the pressure difference between outside and inside the gas bellows will be zero under any operating condition of the system. Gas storage bellows with low spring rates and large effective areas are now applicable. However, the sum of the spring rates of both bellows has to be large enough to hold the pressure of the auxiliary liquid which will be filled into the reservoir under ambient conditions. This remaining spring rate is the reason for a considerable pressure increase within the liquid bellows when the source temperature increases. Since the characteristics of bellows are such that they can stand a higher external than internal pressure, the small bellows is, in the presented design, used in a way that the liquid pressure acts from the outside.

ESTIMATION OF DESIGN PARAMETERS

Reference 1 gives a general steady-state analysis of a feedback-controlled, variable conductance heat pipe. The model describes in differential form the variations of heat-source temperature T_s as affected by changes in heat input Q , the heat sink temperature T_o , and other independent variables. To define the model, a system of five linear differential equations was required.

A general prediction of performance is difficult because, in many applications, changes in each of the independent variables will probably occur. However, a simple estimation of the most important design data, i.e., auxiliary liquid volume and bellows dimensions, is possible with given performance requirements and by assuming constant sink condition.

The equations are derived under the well-known assumptions of the flat-front theory of gas-loaded heat pipes.³ Furthermore, the experimental apparatus is treated as a three-temperature system where T_s is the temperature of the heat source; T_o , the temperature of the heat sink and of the inactive portion of the condenser (T_o is considered to be constant during heat-pipe operation); T_v , the temperatures of the evaporator, of the transport section, of the active portion of the condenser, of the control gas stored in the bellows, and of the feed tube between condenser and bellows.

With these assumptions the governing equations are:

Condition at heat input:

$$Q = (T_s - T_v)/r \quad (1)$$

Condition at heat sink:

$$Q = y(T_v - T_o)/r_c \quad (2)$$

Gas inventory:

$$n = [p(T_v) - p(T_o)]/R \cdot [A_c(L_c - y)/T_o + (V_o + A_2x)/T_v] \quad (3)$$

in which

y	effective condenser length
A_c	vapor cross section of condenser tube
L_c	length of condenser
T_v	vapor temperature
V_o	volume of the feed tube plus rest volume of the bellows at $x = 0$
x	extension of bellows system
A_2	effective area of gas bellows
$p(T_v), p(T_o)$	vapor pressure at T_v and T_o , respectively
R	gas constant
r	thermal resistance between heat source and heat pipe
r_c	external condenser resistance per unit length

It should be pointed out that the heat pipe used in the experimental apparatus was taken from an earlier experiment⁴ and was therefore not specially designed for this test.

The specifications of the heat pipe are as follows:

Dimensions:

Length of evaporator zone: 31.75 cm (12.5 in.)
 Length of condenser zone: 26.7 cm (10.5 in.)
 Length of transport zone: 8.9 cm (3.5 in.)
 Outer diameter: 1.27 cm (0.5 in.) with 0.11 cm (0.042 in.) wall thickness

The cylindrical heat pipe is U-shaped, with the evaporator and the condenser sections as the two legs.

Capillary structure:

Slab wick for the axial liquid flow, with 3.8 mm thickness consisting of six layers of metal felt; circumferential grooves for the radial liquid flow, consisting of 39 threads/cm (100 TPI).

A cooling jacket connected to a constant temperature bath served as heat sink. The external condenser resistance was obtained by providing a

helium-filled annular gap of 2 mm between the condenser wall and cooling jacket. This external resistance was experimentally determined in earlier heat-pipe tests to be $r_c = 15^\circ\text{C cm/W}$.

The chosen performance requirements are:

Sink condition: $T_o = 0^\circ\text{C} = \text{const}$ (32°F)
 Minimum power input condition: $Q_{\min} = 5 \text{ W}$, $T_{s,\min} = 30^\circ\text{C}$ (86°F)
 Maximum power input condition: $Q_{\max} = 30 \text{ W}$, $T_{s,\max} = 33^\circ\text{C}$ (91.4°F)
 Thermal resistance between heat source and heat pipe: $r = 0.5^\circ\text{C/W}$
 Heat-pipe working fluid: methanol

Substituting the maximum and minimum power conditions in Eq. (1) and solving for T_v we obtain $T_{v,\min} = 27.5^\circ\text{C}$; $T_{v,\max} = 18^\circ\text{C}$.

Equation (2) gives the required condenser length y for $T_o = \text{const}$: $y_{\min} = 2.7 \text{ cm}$; $y_{\max} = 25 \text{ cm}$.

As seen in Eq. (3), the gas volume is divided into three parts:

$A_c(L_c - y)$ gas volume within the condenser region

V_o volume of the feed tube plus rest volume of the bellows at $x = 0$

A_2x change in gas volume due to movement of bellows

The bellows system will be preferably designed with $x = 0$ at the minimum power input condition. For this case, Eq. (3) yields a relation between n and V_o (with a given condenser geometry). As shown later, the rest volume of the bellows is about 70 cm^3 , which is undesirably large. Adding 10 cm^3 for the feed-tube volume, V_o , yields 80 cm^3 . With this value we obtain, from Eq. (3), the molar gas inventory:

$$n = 5.59 \times 10^{-4} \text{ mole}$$

By entering this value in Eq. (3) for the maximum power condition, we find the required maximum gas volume within the bellows:

$$A_2x = 77.4 \text{ cm}^3 \quad (4)$$

The effective extension of x of the bellows is equal to the difference between the extension due to the thermal expansion of the liquid x_t and the compression caused by the pressure Δp within the slightly compressible liquid x_p :

$$x = x_t - x_p$$

in which

$$x_t = \frac{\beta V_l \Delta T_s}{A_1} \quad (5)$$

$$x_p = \frac{\alpha \cdot V_l \Delta p}{A_1} \quad (6)$$

with

β coefficient of thermal expansion

α coefficient of compressibility

A_1 effective area of liquid bellows

V_ℓ volume of auxiliary liquid

Δp change in pressure of auxiliary liquid

The extension x can also be expressed using the spring rate, $k = k_1 + k_2$ of the bellows system:

$$x = \frac{A_1}{k} \cdot \Delta p \quad (7)$$

Equations (5), (6), and (7) lead to

$$\frac{x}{\Delta T_s} = \frac{\beta}{A_1/V_\ell + k \cdot \alpha/A_1} \quad (8)$$

and combining Eqs. (4) and (8) finally gives the relationship between the effective areas of the bellows and the required auxiliary liquid volume V_ℓ with given liquid properties and by considering $\Delta T_s = 3^\circ\text{C}$

$$V_\ell = \frac{A_1^2}{3.87 \times 10^{-2} A_1 A_2 \beta - k \alpha} \quad (9)$$

SELECTION OF AUXILIARY LIQUID

To obtain a large extension of the bellows system, β has to be large and α small. Fortunately, the compressibility factor for most liquids is about two orders of magnitude smaller than their thermal expansion coefficient. The fact that α appears in Eq. (8) only in a sum term decreases its influence further.

To prevent high pressure in the auxiliary liquid, its boiling temperature should be equal to or higher than the heat-source temperature of the heat-pipe system. The relationship between thermal expansion and critical or boiling point of a liquid gives the law for the thermal expansion of normal liquids by Duggar⁵:

$$(1/V) (\Delta V/\Delta T) = 0.3/(T_c - T)$$

in which

V volume of the liquid

T temperature of the liquid, $^\circ\text{K}$

T_c critical temperature of the liquid, $^\circ\text{K}$

and the finding by Benke⁶ that the ratio of the boiling and critical temperatures is constant for many compounds.

Accordingly, the thermal expansion is high for liquids with low critical or boiling temperatures. This indicates that a passive feedback system might be especially effective in the cryogenic region. In fact, the expansion coefficient of liquid nitrogen, for example, is three to four times higher than values for liquids at ambient temperatures.

Table I contains the β and α values of some selected liquids. The data, except those for $n - C_5F_{12}$, were calculated from relative volume values given in Refs. 8 and 9. The expansion of the liquids is almost linear in the indicated temperature range, and the error in applying these values for the small temperature change in this application should be small. The smallest pressure step for the calculation of the α values is 500 atm. In this region, the compressibility curve has a somewhat hyperbolic slope, with higher α values at lower pressures. The values in the 1 to 10 atm pressure range, which is interesting for this application, are actually higher than the average values given in Table I. Relative volumes in the lower pressure range were found only for iso-pentane.¹⁰ By taking these values at 5.44 and 10.88 atm, α is calculated to be $2.52 \cdot 10^{-5} \text{ cm}^2/\text{N}$, which is twice the value given in Table I. The values in Table I for ether, acetone, and methanol probably need similar adjustments prior to their application in the lower pressure range.

Since perfluoro-n-pentane has the highest β value, this liquid was chosen for the presented heat-pipe system.

EXPERIMENTAL APPARATUS

Cross sections of the experimental apparatus are shown in Figs. 3 and 4. The two basic components of the experiment are primarily the integral part, consisting of the heat source, the auxiliary liquid reservoir, the heat-pipe panel, and the bellows system and, secondly, the heat sink, which consists of the condenser section of the heat pipe surrounded by a cooling jacket.

To obtain a large effective area, a bellows with rectangular cross section as gas reservoir was first considered. A rectangular cross section would ideally fit a rectangular-shaped heat source, which is found in many applications. However, discussions with several bellows companies indicated that, besides the high price, the manufacturers would not be able to produce such bellows with a usable low spring rate. As a result, two cylindrical, relatively low-priced, "off-the-shelf" bellows, actuated by one auxiliary liquid bellows, were chosen. The liquid bellows is separate from the gas bellows and joined to them by a mechanical removable connection. This makes it possible to test and, if desired, to replace the auxiliary liquid system independently from the gas bellows system.

The specifications of the selected bellows, as given by the manufacturers, are as follows:

	Auxiliary Liquid Bellows	Gas Bellows
Type	05*	350-250**
No. of segments	5	3
Effective area, cm ²	0.316	45.55
O.D., cm	0.953	8.9
Free length, cm	2.5	2.44
Length in compression, cm	1.27	1.22
Length in extension, cm	3.56	4.88
Spring rate, N/cm	7.06	10.6
Material	AISI 347	AISI 347
Spring rate of entire bellows system		28.26 N/cm
Effective area of the two gas bellows		91.10 cm ²

Considering perfluoro-n-pentane as auxiliary liquid, Eq. (9) yields directly the required volume of the liquid reservoir:

$$V_L = 72.4 \text{ cm}^3$$

It is emphasized that no attempt was made to optimize the bellows system other than to select a readily available bellows that would provide sufficient variation of the gas reservoir to obtain the desired control in the chosen power range. Especially, the ratio between the size of the bellows and the auxiliary liquid volume depends on the specifications of a given application. Furthermore, for a more accurate system, AM 350 steel bellows are suggested. These custom-made bellows can be heat treated in any position of the bellows so that the most convenient free length can be obtained. To use, for instance, a gas bellows, which is compressed in the load-free condition, will reduce the dead volume V_0 (i.e., Eq. (3)) and the overall height of the bellows system. To minimize V_0 in the present design, a hollow displacement body is inserted in each bellows. The bellows system is covered by the already mentioned container, which contains a small amount of heat-pipe liquid (methanol).

The heat source consists of an aluminum block, which is heated from underneath by means of three foil heaters. Since the selected auxiliary liquid ($n\text{-C}_5\text{F}_{12}$) has a low thermal conductance (about 3000 times smaller than aluminum), it was necessary to provide a reservoir with a large wall surface, which would reduce the time needed to adjust the liquid temperature to that of the heat source. This was accomplished by drilling 18 holes of 6.35 mm diameter into the aluminum block (17 holes perpendicular to the larger side and 1 hole perpendicular to the smaller one). All holes, except the filling port, were sealed by using common pipe plugs and an epoxy resin.

The thermal resistance was imposed by inserting a 3.5-mm-thick nylon sheet between the heat source and the heat-pipe panel. The panel consisted of two copper plates in which two mating grooves were machined to accommodate the heat pipe when the plates were clamped together. To reduce the heat resistance between the two plates, all mating surfaces were covered with a layer of thermal grease prior to mounting.

*Metal Bellows Company, Chatsworth, California.

**Standard Bellows Company, Windsor Locks, Connecticut.

TEST PROCEDURE

Prior to testing, some preparatory procedures were necessary, which are briefly presented.

Determination of V_ℓ and V_o

Before the test apparatus was assembled, the auxiliary liquid reservoir and the rest volume of the bellows and the feed tube were experimentally determined to be $V_\ell = 68$ cm and $V_o = 82$ cm. This was done by evacuating and refilling with a measured amount of methanol.

Charging the Heat Pipe

The heat pipe was filled with helium as the control gas and methanol as the working fluid. The gas was put in first at a pressure of 1.24 N/cm^2 (1.8 psi) at room temperature, which corresponds to approximately 5.6×10^{-4} mole. Then the pipe was filled with 22 cm^3 methanol at room temperature. During this fill procedure, the pressure within the container (Fig. 4) was still atmospheric, which held the bellows in the most closed position. The heat pipe was now ready for the first test run without feedback control.

Calibration of the Linear Transducer

An AC-AC linear transducer, which operates as half of a 4-arm AC bridge circuit, was used to monitor the position of the bellows system. A carrier amplifier served as the reference half of the bridge. Moving the probe away from the null point causes an output voltage across the bridge, which was measured with a precision digital voltmeter. Prior to installation, the transducer was calibrated with a micrometer. The calibration curve confirmed the linearity of the output voltage with probe displacement within a useful range, as specified by the manufacturer.*

Charging the Container

After installation of the transducer, the container was clamped to the heat-pipe panel, evacuated, and then filled with 10 cm^3 methanol. The evacuation caused the bellows to extend to the container wall, which served as an extension stop. Inserting the methanol did not affect the bellows positions since the pressure of the methanol vapor-helium mixture within the heat pipe and bellows was still higher than the pressure of the pure methanol vapor within the container. Only during start-up of the heat pipe, when the helium begins to separate from the vapor of the working fluid, the bellows start to compress because of their own spring rate.

Charging the Auxiliary Liquid Reservoir

The effort to fill the liquid reservoir turned out to be the most critical procedure of the test program. The first attempt was made with the gas bellows in the most compressed position (prior to filling the container). However, a subsequent heat-pipe run failed to show any movement of the bellows.

*Moxon Electronics, SRC Division, Sunnyvale, California.

Obviously, the reservoir was not completely filled with liquid. The next step was to fill the reservoir with the gas bellows in the extended position (after filling the container) with the aim to pressurize the liquid during movement of the bellows to their closed position. This was a rather dangerous method. If the reservoir had been filled completely at this time, an increase in the source temperature and the consequent expansion of the liquid would have caused a damaging overpressure in the liquid bellows, which was already located against the extension stop. But the contrary happened: during heat-pipe start-up, the gas bellows moved to the most compressed position as if the reservoir contained no liquid at all.

To overcome this problem, a device to pressurize the liquid after filling was added to the system. This consisted of a simple brass piston and cylinder, sealed by an O-ring. After starting and operating the heat pipe at minimum power, the piston was actuated until the bellows began to move. Once this happened, the reservoir was sealed off and the system was ready for the proposed test run.

Since Depew et al.² had similar difficulties in their tests with a passive feedback-controlled heat pipe, such a device to prepressurize the liquid is apparently necessary and should be included in the design of a passive feedback system. It is suggested that another small bellows be used for this purpose rather than devices that are susceptible to leaks. This bellows may be actuated by a screw mechanism and the entire assembly could be integrated in the liquid reservoir. Since it is possible to vary the position of the bellows system by actuating the piston, with a subsequent change in heat-pipe pressure, such a device may be, in addition, used to adjust or, in certain limits, to change the operation temperature of the system.

EXPERIMENTAL RESULTS AND DISCUSSION

The test program was divided into three parts: (1) gas bellows held in the most compressed position, (2) feedback control and methanol as auxiliary liquid, and (3) feedback control and n-perfluoro pentane as auxiliary liquid.

The primary results are shown in Figs. 5, 6, and 7. Both runs with feedback control show superior source temperature control compared to the test with the bellows held in closed position. In addition, the evaporator temperature decreases as predicted, when the power input increases, a feature which makes a close control of the source temperature possible and which cannot be produced with conventional gas-loaded heat pipes. A comparison between the runs with the two different auxiliary liquids shows the expected better control performance with perfluoro pentane.

The effective control of source temperature obtained with perfluoro pentane is $\Delta T_s \approx 3^\circ\text{C}$, over a power range between 3 and 30 W. The corresponding decrease in the evaporator temperature, however, is only $\Delta T_{ev} = 5.5^\circ\text{C}$, compared to the predicted 11°C . The reason for this difference is a lower than expected thermal resistance between heat source and heat pipe ($r = 0.31^\circ\text{C/W}$ rather than 0.5°C/W). However, with this smaller value, one would still expect a better control than $\Delta T_s = 3^\circ\text{C}$. The reduced control is explained by comparing the predicted and the experimentally determined bellows movement. Equation (8) yields $x/\Delta T_s = 2.73 \text{ mm}/^\circ\text{C}$ for perfluoro pentane, while the experimental results indicate only $x/\Delta T_s = 2.34 \text{ mm}/^\circ\text{C}$, or 14% less than predicted. Although the reservoir was helium-leak checked, small amounts of

liquid could apparently escape. This was confirmed by the observation that the bellows system was not able to hold the initial position during a 24-hr test at constant power level.

CONCLUSIONS

The test program has demonstrated the feasibility of the passive feedback control system and its superior control capability compared to conventional gas-loaded heat pipes. Between 3- and 30-W power input, the change of the source temperature was determined to be $T_s = 3^\circ\text{C}$ with feedback control compared to $T_s = 11^\circ\text{C}$ with the bellows held in closed position. The concept to integrate the heat source, the heat pipe panel, and the bellows system was successful. The fact that there existed no significant pressure difference between the inside and outside of the gas bellows led to a more convenient and consistent operation. The concept presented is especially applicable in connection with large, flat, heat-source surfaces. A considerable reduction in weight is possible by using a flat heat-pipe evaporator rather than a solid panel, which is cooled by a cylindrical heat pipe. Further improvements in control capability can be obtained by minimizing the dead volume of the gas reservoir (V_o) by utilizing bellows, which are compressed in the load-free condition.

Work is now in progress to further optimize the system.

ACKNOWLEDGMENTS

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TABLE I: β AND α VALUES OF SOME SELECTED LIQUIDS

Liquid Formula	Perfluoro- n-Pentane $n-C_5F_{12}$	Ether C_4H_8O	Iso-Pentane C_5H_{12}	Acetone C_3H_6O	Methanol CH_3OH
$\beta \times 10^3$ [$^{\circ}C$]	2.06	1.72	1.69	1.48	1.2
Temp. range [$^{\circ}C$]	at 25	20-40	0-50	20-40	20-40
$\alpha \times 10^5$ [cm^2/N]	3.24	1.28	1.25	0.9	0.85
Pressure range [atm]		1-500	1-500	1-500	1-500
Boiling Point [$^{\circ}C$]	29.3	34.6	27.9	56.2	65.0
Reference	7	8	9	8	8

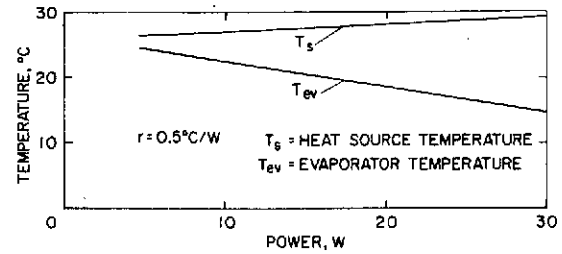
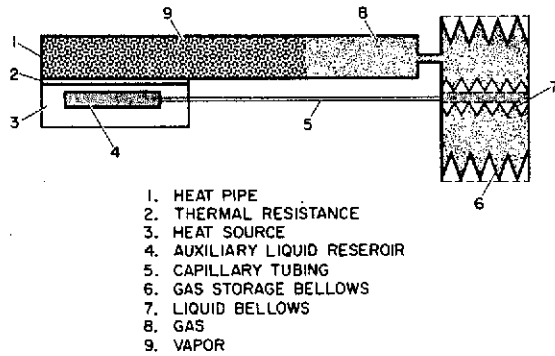


Fig. 1: Schematic of feedback control system.

Fig. 2: Desired performance.

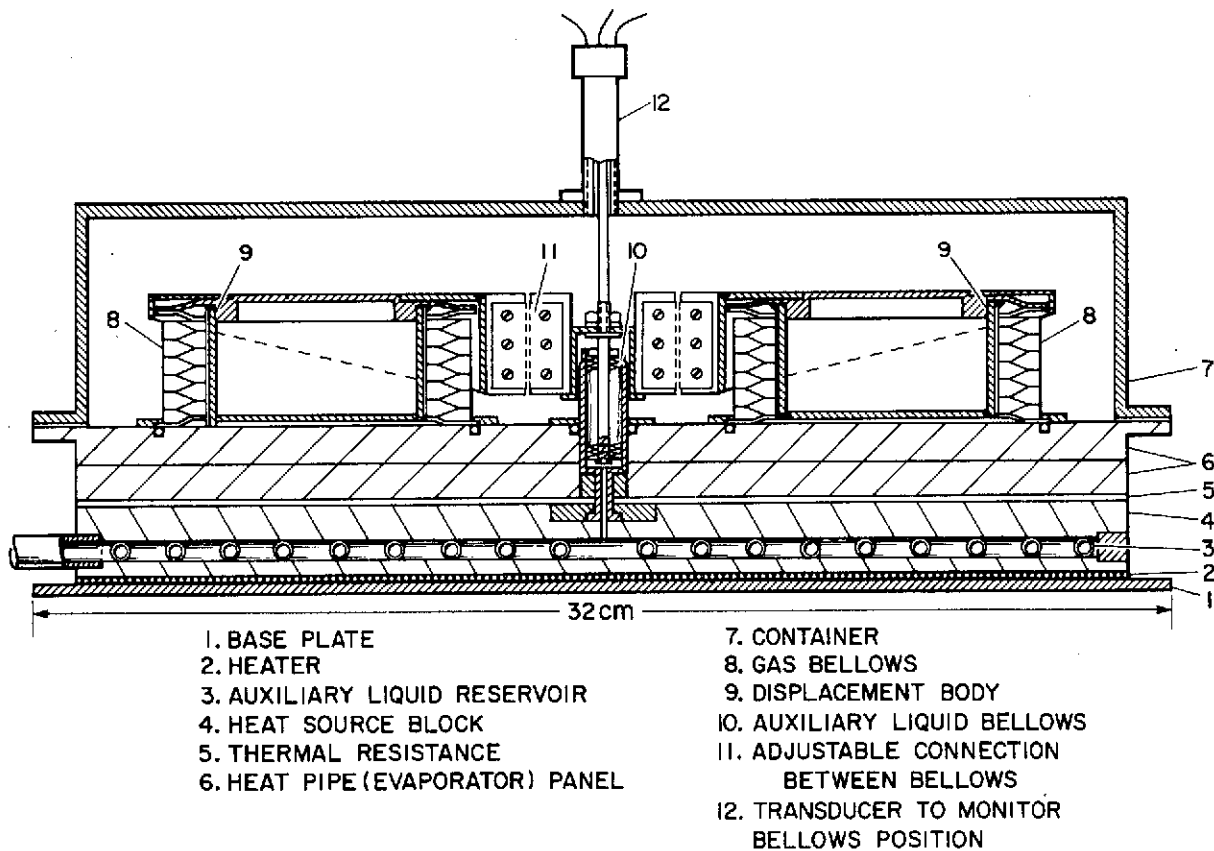


Fig. 3: Cross section of the experimental apparatus.

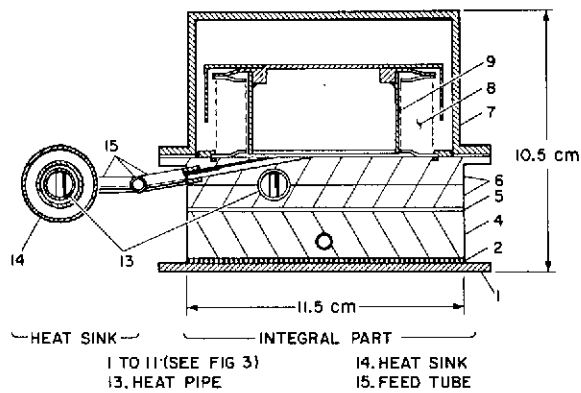


Fig. 4: Cross section of the experimental apparatus.

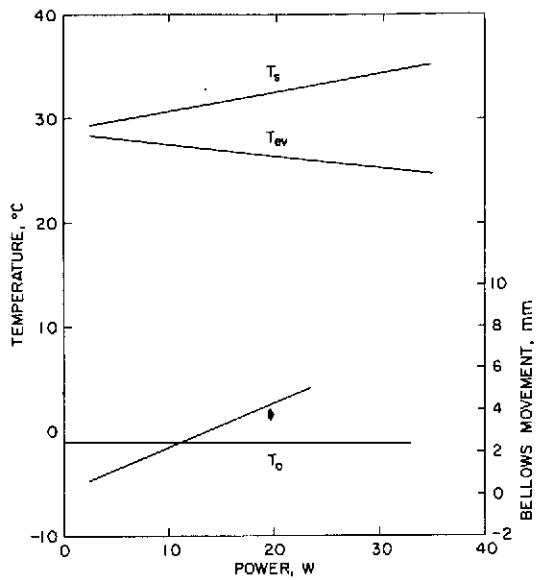


Fig. 5: Source temperature T_s , evaporator temperature T_{ev} and bellows movement vs. power with methanol as auxiliary liquid.

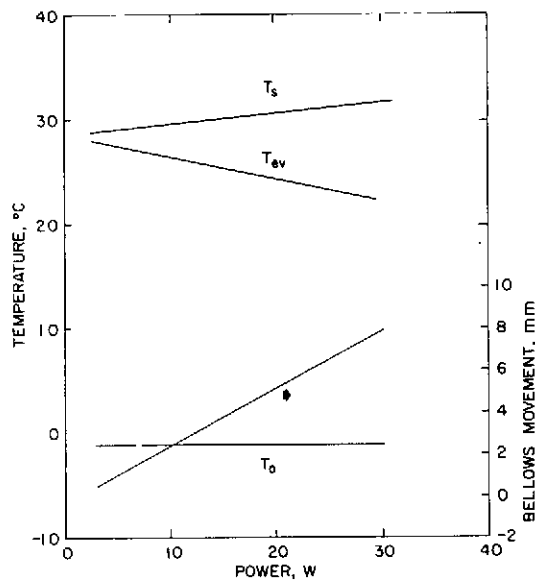


Fig. 6: Source temperature T_s , evaporator temperature T_{ev} and bellows movement vs. power with n-perfluoro-pentane as auxiliary liquid.

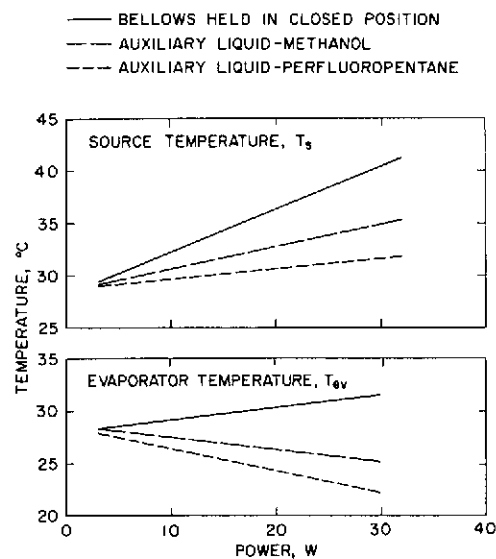


Fig. 7: Comparison of control capability when different auxiliary liquids are used and when the bellows are held in the closed position.